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Lewis Research Center
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TECHNICAL PAPER proposed for presentation at
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by H. W. Scibbe, D. E. Brewe, and H. H. Coe

Lewis Research Center
Cleveland, Ohio

ABSTRACT

Rolling element bearings used in liquid hydrogen turbopumps have lubrication, material, and design requirements somewhat different from those used in high-speed oil lubricated applications. The bearing load-carrying surfaces are lubricated by transfer films provided from a self-lubricating cage material. The bearings are designed for minimum heat generation at high speed by using open race curvatures and small diameter balls. In addition to the self-lubricating feature, the cage must have adequate strength and be designed with proper clearances to prevent seizure or excessive wear in the cold environment that can result in premature bearing failure.

With proper materials and design 40 mm bore ball bearings have operated successfully in -400° F hydrogen gas at 20 000 rpm. Cage wear with four Teflon-compound materials was less than 0.25 percent of the original cage weight for running times up to 10 hours.

INTRODUCTION

During the past decade technology has been developed for rolling-element bearings used in cryogenic applications. In cryogenic fluids such as liquid hydrogen (-423° F, boiling point), liquid oxygen (-297° F), and liquid nitrogen (-320° F) ordinary oils and greases cannot be used as bearing lubricants, since they become glass-like solids. The lubricating mechanism for bearings operating in cryogenic fluids consists primarily in the provision of low-shear strength films or oxides on the bearing surfaces to maintain integrity and prevent welding. Operation in liquid hydrogen is the most difficult because initial oxide films present on the bearing surfaces cannot be reformed in the reducing environment once they have been worn away. Lubrication must therefore be provided solely by transfer films from a solid, self-lubricating cage material.

Fulfillment of the operating requirements of rolling-element bearings used in liquid hydrogen turbopumps has contributed significantly toward the development of design technology used in all cryogenic bearings. Ball bearings used in hydrogen turbopumps usually operate at high rotative speeds, under moderate thrust loads for short time periods. Although the turbopump operates for only a few minutes (the duration of the rocket engine firing), a high degree of reliability is required by all its components, including the bearings, to preclude failure of an expensive and complex rocket engine system. Total bearing run time is usually 4 to 5 hours including system checkout and engine static firing tests.

The purpose of this paper is to present several basic design and material requirements of ball bearings used in liquid hydrogen turbopumps. Among these are the selection of: (1) an internal design with geometric factors that minimize heat generation within the bearing due to ball spin; (2) a ball-race material with good dimensional stability and adequate low-temperature ductility; (3) a self-lubricating cage material that provides lubricant on the bearing load carrying surfaces and has a long wear-life; and (4) a cage design with adequate strength and sufficient clearances at the operating cryogenic temperatures.

Most of the information presented on bearing materials and design was obtained from experimental bearing programs conducted in cryogenic hydrogen (both liquid and cold gas) at the Lewis Research Center during the last ten-year period. Specifically, the ball bearing internal design for minimum heat generation was obtained from a program where the bearings operated at high speeds submerged in a liquid hydrogen bath.(1) The cage material and design information was obtained from an experimental bearing program run in -400° F gaseous hydrogen.(2, 3) It was anticipated, that at high rotative speeds, operation in cold gaseous hydrogen subjects the bearings to more severe operating conditions than does liquid hydrogen. These conditions result in higher cage wear and lower cooling capacity.

Ball and race material selection was based on experiments conducted in liquid nitrogen and hydrogen. In these runs the bearings were operated for several hundred hours.(4)

Internal Bearing Design for Minimum Heat Generation

A primary cause of failures in high-speed ball bearings is excessive heat generation. The major sources of heat generation in the bearings are the ball spinning friction in the ball-race contact areas and the rubbing friction between the cage and balls and the cage locating surface on one of the races. The heat generated within the bearing must be removed to assure an equilibrium operating condition and to prevent a total loss in operating clearance that results in seizure. A heat balance within the bearing can be maintained by designing for minimum heat generation and then removing the heat generated with the lubricant or coolant. Although liquid hydrogen has excellent coolant properties it is a poor lubricant. The problem of maintaining a heat balance within high-speed bearings operating in cryogenic hydrogen is therefore more difficult than with oil lubrication. Furthermore, because of higher friction coefficients in hydrogen[5] between surfaces in sliding contact, frictional heat is developed within the bearings at a greater rate.

Selection of a bearing design for minimum heat generation. - As previously stated, one major source of heat generation in ball bearings is the spinning which occurs in the contact between the balls and one race. Spinning occurs between the balls and one race in all ball bearings that operate under a thrust or combined thrust and radial loads. Ball spinning increases during high speed operation due to a change in contact angle. As shown in figure 1, a bearing under thrust load P operates at some contact angle, β. At low speeds, the contact angles β are equal at the inner- and outer-race contacts

(fig. 1(a)). When the bearing is operated at high speed, ball centrifugal force creates an additional load at the outer-race contact which results in unequal contact angles at the inner- and outer-race contacts (fig. 1(b)). The magnitude of this difference in operating contact angles further increases ball spinning and consequently more heat is generated by the bearing.

One solution to problems of ball centrifugal force at high rotative speeds is to use smaller balls. The load capacity of a ball bearing operating under thrust load is proportional to the product of the number of balls (n) and the square of the ball diameter (d^2). In order that the load capacity not be appreciably reduced when decreasing the ball diameter, it is necessary therefore to increase the number of balls in the bearing. There is a practical limit for reducing ball diameter or increasing the number of balls and that is the cage strength, especially between adjacent ball pockets. Also if the cage annular cross-section is reduced excessively it may break at high speed because of inadequate strength.

An analysis was made and programmed on a high-speed digital computer to determine the design and geometric factors that influence heat generation rates in a high-speed ball bearing operating in liquid hydrogen. The computer results which were supported with experimental data[(1)] indicated that the following factors contribute to lower heat generation rates: (1) open race curvatures (i.e., large value for the ratio of race groove radius to ball diameter), (2) small ball diameters, and (3) ball spinning at the race with larger curvature.

The results of the computer program for 40-millimeter bore ball bearings with two different ball diameters and using two race curvature combinations are shown in table I. The heat generated due to ball spin is converted to shaft torque. The torque values shown are for ball spin at the inner-race contact. The 108 series bearings have larger torque values for both race curvature combinations due to the greater heat generated by the larger ball size. Increasing the race curvature from 0.51 to 0.52 at the spinning contact resulted in a decrease in torque for both size bearings. It should be noted that the large decrease in torque from the 108 bearing with an inner-race curvature of 0.51 to the 1908 bearing with a 0.52 inner-race curvature is a function of both ball size and race curvature.

Another example where increasing the race curvature at the spinning contact resulted in decreased ball-spin torque is illustrated in the turbopump bearings for the NERVA (Nuclear Engine for Rocket Vehicle Application) engine. The pump ball bearings are 50-millimeter bore angular contact type (210 series), and operate at a design speed of 24 000 rpm and a rated thrust load of 2000 pounds.[6] A computer analysis, similar to that described above, was made for these 50-millimeter ball bearings to indicate the effect of several race curvature combinations on ball-spin torque. Ball-spin torque, occurring at the outer-race contact, is plotted in figure 2 for shaft speed from 15 000 to 30 000 rpm. Increasing the outer-race curvature from 0.52 to 0.54 decreased the torque approximately 40 percent throughout the speed range. Increasing the outer-race curvature to 0.58 decreased

the torque approximately 70 percent from the 0.52 curve. An effect of ball centrifugal force on the outer-race contact can be seen in the lowest curve. The ball spinning changes from outer race to inner race at approximately 27 000 rpm. This transition occurs because at the higher rotative speeds, ball centrifugal force increases and the balls grip the 0.58 outer-race contact more firmly. Ball spinning, therefore, occurs at the 0.54 inner-race contact and considerably more heat is generated. This transition from spinning at the outer-race contact to spinning at the inner-race contact would not occur as long as the shaft speed of 24 000 rpm is not appreciably exceeded. In the NERVA turbopump bearing test program, several bearings with 0.52 outer - 0.52 inner-race curvatures and the 0.54 outer - 0.58 inner-race curvatures have been run in liquid hydrogen at 24 000 rpm at thrust loads to 2000 pounds, for operating times up to 90 minutes without failure or gross cage wear.

Ball-Race Materials

In addition to the high compressive strength and hardness requirements necessary in bearing steels, properties such as corrosion resistance, dimensional stability, and low-temperature ductility are required for ball and race materials used at cryogenic temperatures. Conventional SAE 52100 bearing steel and AISI 440C stainless steel have both been used successfully as ball and race materials in cryogenic bearings applications.[1, 4, 7] The 440C stainless steel exhibits properties similar to those of 52100 steel at these low temperatures, but is generally preferred because of its better corrosion resistance.

Dimensional instability in bearing steels, such as 440C stainless, is the result of appreciable amounts (10 to 15 percent by volume) of retained austenite in the crystalline structure after heat treatment. Conversion of the unstable austenite to martensite can result in an increase in physical bearing dimensions. The transformation to martensite is delayed and usually occurs during bearing service. This is particularly true when the bearings are operated at temperatures below that of the quenching bath, such as in cryogenic applications. To convert the retained austenite to martensite the steel is subcooled to -100° F or lower, immediately after quenching. Subcooling is followed by tempering to stabilize the newly formed martensite. To insure maximum dimensional stability in 440C stainless and other bearing steels that will be used in cryogenic applications, subcooling to temperatures as low as -300° F is specified as part of the heat treat cycle.

Although 440C stainless steel exhibits an increase in tensile (compressive) strength and hardness at cryogenic temperatures, it also experiences an increase in brittleness and a decrease in impact strength. Specific data on the low-temperature ductility of 440C is not readily available, however it may be assumed that the balls and races do retain some ductility, even at temperatures as low as liquid hydrogen (-423° F). Ring cracking that has been observed in several cryogenic bearing applications,[7] was probably a result of excessive shrink fits between the 440C races and the shaft or housing materials. The results of preliminary experiments, with the bearings operating

immersed in liquid nitrogen and hydrogen, indicated that the differential contraction rates of the various materials used can, upon cooling, affect bearing running clearance.[1] The ball-race materials, the bearing mounting materials, and the mounting clearances therefore should be selected so as to minimize the reduction in bearing diametral clearance at the cryogenic temperature. Additionally, the cage clearances should be selected sufficiently large to account for the differential contraction between the cage and the ball-race materials at cryogenic temperature. (Cage materials, design, and clearances are discussed thoroughly in subsequent sections.)

The measured, room-temperature, diametral and cage clearances in hydrogen turbopump bearings are usually two to three times those required in high-speed oil lubricated bearings. The actual clearances specified are largely dependent upon the bearing size, the maximum operating speed, and the relative contraction rates between the ball-race and cage materials as the turbopump assembly is cooled to liquid hydrogen temperature (-423° F).

Since the required life of bearings operating in cryogenic turbopumps is only a few hours, rolling element fatigue is not considered a major cause of failure. It is generally recommended however, that the maximum Hertz compressive stress in the ball-race contacts be limited to 350 000 pounds per square inch.

Bearing Lubrication by Transfer Films

In ball bearings using a conventional lubrication system, surface integrity is maintained and surface welding prevented by the presence

of contaminant surface films or by a lubricant that separates the surfaces in sliding and rolling contact. In cryogenic hydrogen lubrication of the bearing load-carrying surfaces is provided by transfer films from a self-lubricating cage material. The ball-race contacts are the most critical areas requiring lubrication, because of the high contact stresses. A film transfer process through which the bearing surfaces are lubricated is illustrated in figure 3. As the bearing rotates the balls rub in the cage pockets. Thin films of cage material or lubricant are transferred to the balls and subsequently by the balls to the race grooves. The cage locating surface on one of the races is lubricated directly by contact with the cage material.

Self-lubricating cage materials. - Attempts to operate ball bearings in cryogenic hydrogen with cage materials normally used in oil lubrication resulted in failure. Two of these failures are shown in figure 4. In figure 4(a) a 40-millimeter ball bearing with a cotton-cloth phenolic cage showed extreme ball-pocket wear after operating only 80 minutes in liquid hydrogen. In the other application (fig. 4(b)) a turbine flowmeter, using miniature size (0.047 inch bore) ball bearings, failed after 90 minutes operation in liquid hydrogen. These bearings were equipped with steel cages using the "L" type design. These two failures illustrate the fact that the cage materials must be self-lubricating to assure successful operation in cryogenic hydrogen.

The best lubricant used to date for cryogenic bearing cages has been Teflon (polytetrafluoroethylene). Teflon provides low friction at the bearing sliding contacts, but it cannot be used in its pure form as

a cage material because of poor strength properties and because of its tendency to cold flow even under the lightest loads. Teflon also has poor thermal conductivity which becomes a problem at high cage speeds, where heat generation in the bearing may become detrimental to successful operation. Teflon must therefore be compounded with other materials (fillers) to give it these desirable properties, and provide a good wear life as a cage material. Glass fiber fillers greatly improve the mechanical strength of Teflon. Additions of molybdenum disulfide (MoS_2) increase the hardness of the Teflon material. An increase in material hardness usually improves its wear resistance. Generally, MoS_2 is added in small percentages and is used in conjunction with other fillers, such as glass fibers or bronze. Materials with bronze fillers showed higher hardness with improved thermal conductivity and strength above that of pure Teflon. The improvements in mechanical and physical properties provided by these and other fillers in Teflon are described in more detail in [8].

The structures of four Teflon compounded materials are illustrated in figure 5. Figure 5(a) shows a laminated structure consisting of alternate layers of glass cloth and Teflon binder. The material is 38-percent (by weight) glass cloth and 62-percent Teflon. The glass appears as rod-shaped fibers bundled together to form strands. The strands are then woven into a continuous cloth layer. Figure 5(b) shows a 15-percent-glass-fiber, 5-percent-MoS_2, and 80-percent Telfon material with rod-shaped glass fibers and large particle-size (approximately 100 microns) MoS_2 in a Teflon matrix. Figure 5(c)

shows a 15-percent glass-fiber and 85-percent Teflon material composed of rod-shaped glass fibers in a Teflon matrix. In figure 5(d) a 30-percent bronze and 70-percent Teflon material is illustrated. The bronze particles, also approximately 100 microns in size, are dispersed in a Teflon matrix.

The four Teflon materials were evaluated, along with three other materials, as cages in 40-millimeter ball bearings. The bearings were operated in -400° F gaseous hydrogen at 20 000 rpm and 200-pound thrust load for running times up to 10 hours.[2, 3] Transverse profile traces were made on the inner-race grooves to measure the film buildup from transferred cage material and determine the extent of race wear.

Measurement of transfer films. - A profile tracing technique was used to study the formation and life histories of the transfer films provided from the cages on the bearing inner races. A cross-section of the bearing inner-race groove and a typical profile trace are shown in figure 6. As the stylus traces the race groove contour, a highly magnified profile trace is produced as shown in the lower portion of figure 6. Of particular interest is the ball-track region where lubrication is required. The horizontal line shown on the profile trace is the original race groove contour. The area of profile trace above the line is transfer film (lubricant) and the shaded area below the line is race wear. Successive profile traces were made at intervals of approximately 2 hours running time to study the history of the transfer films.

<u>Laminated-glass-cloth-with-Teflon-binder cage material (38 percent glass cloth, 62-percent Teflon)</u>. - This material, currently used in several turbopump bearings, is quite strong because of its laminated structure. Typical inner-race profile traces of a bearing using this cage material are shown in figure 7. Illustrated is the life history of the transfer film for approximately ten hours total running time. After running for 284 minutes (fig. 7(c)) a fairly good film of lubricant has been deposited on the inner race. The scratches that appear in the film are indicative of abrasion caused by the glass in the cage material. With continued running, the film breaks down and wear of the inner race surface begins (fig. 7(d)). Race wear progresses with time and, after 10 hours running, the bearing is in danger of failing.

The cause of the film breakdown and the large amount of race wear is probably abrasion by the glass fibers which are shredded from the cage material. This postulated abrasive-wear process is illustrated in figure 8. Shown are alternate layers of glass cloth and Teflon. The ball rotates and wears away the softer Telfon material. The Teflon is deposited in the race groove as the lubricant film. After a short time the Teflon has worn back leaving the glass cloth exposed to the rubbing action of the balls. Continued running results in film breakdown, caused by glass fibers which are shredded away from the cloth and embedded into the film. The film is worn away faster than it can be reformed by the Teflon and heavy race-groove wear results.

<u>Filled-Teflon cage materials</u>. - Successive profile traces of the inner-race grooves were also made, at approximately 2 hour intervals, for bearings run with cages made from glass-fiber and MoS_2-filled

Teflon, glass-fiber-filled Teflon, and bronze-filled Teflon. The profile traces at the end of each bearing 10-hour run series are shown in figure 9. Although some scratches appear in the films and the glass-fiber-MoS_2-filled-Teflon material (fig. 9(a)) indicates some race wear, the transfer film thickness was adequate to provide good lubrication throughout the entire run series.

Cage wear. - The four Teflon cage materials exhibited extremely low wear for their 10-hour running times in cold hydrogen gas. Percent weight loss for each of these four materials is plotted in figure 10 as a function of total sliding distance at the cage inner diameter. The maximum weight loss was 0.25 percent or less of the original cage weight with the exception of one bearing with a bronze-filled Teflon cage (figure 10(d)). This cage had a weight loss of 0.89 percent. The high wear was attributed to eccentric shape of the cage, which rubbed heavily on the inner-race load.

Other cage materials investigated in -400° F hydrogen gas were a copper-composite, a silver-composite, and a MoS_2-filled polyimide. The composite materials provided good film transfer on the bearing inner-race grooves. The test runs were ended prematurely however, because of high cage wear. The MoS_2-filled polyimide material failed completely after running less than one hour in two evaluation runs.

Bearing Cage Design

Another important consideration for ball bearing cages used in cryogenic hydrogen turbopumps is their design. In addition to the self-lubricating feature, other properties of the cage material such

as mechanical strength, structural rigidity, and wear resistance must be acceptable.

Mechanical strength at cryogenic temperatures must be accompanied with a good strength-to-weight ratio. Materials that do not have sufficient strength by themselves should be reinforced on their outer surfaces with metal shrouds.

Structural rigidity is required because the cage material must be rigid enough to withstand deformation, but not become so brittle at cryogenic temperatures that it fails by cracking.

In addition to the strength and rigidity requirements, the material must have good wear resistance. In order that material wear resistance be used to its full advantage, the cage must be designed with proper ball-pocket and cage-locating (land) clearances.

Inner-race located cage design. - Three designs used successfully for inner-race located cages are shown in figure 11. These designs were used in the cage material evaluation study previously discussed. The 40 millimeter ball bearings were run in -400° F hydrogen gas at 20 000 rpm and 200 pound thrust load. Of the three designs illustrated in figure 11 the one selected for a particular turbopump bearing is usually dependent upon the strength and the strength-to-weight ratio of the cage body materials at the cryogenic temperature.

Outer-race located cage design. - Other cage designs that have been used successfully in the NERVA(6) and other(7) turbopump bearings are shown in figure 12. The conventionally designed cage (fig. 12(a)) has a heavy cross-section with reinforcing side plates, similar

to the inner-race located design shown in figure 11(b). Because of the large rotating mass the cage generates considerable heat at the outer-race locating surface.

The thin-line, lighter weight design shown in figure 12(b) is flexible and sufficiently strong because it depends on the rigidity of the outer race for reinforcement. The open design also permits greater thru-flow of liquid hydrogen and therefore results in better cooling efficiency of the bearing than does the conventional cage design. The laminated-glass-cloth-with-Teflon-binder cage material has been used successfully with the thin-line design in several turbopump bearings.

Cage clearances. - As indicated previously (p. 8), the cage clearances specified for hydrogen turbopump bearings are somewhat dependent upon the relative contraction rates between the ball-race and cage materials, as the bearing is cooled to -423° F. The total linear contraction from 70 to -423° F for 440C stainless steel, ball and race material, is approximately 0.0019 inch per inch[9]. In contrast the contraction for 100 percent Teflon is 0.0215 inch per inch[10] more than an order of magnitude greater over the same temperature range. Filler materials added to Teflon resin (p. 10) will reduce the total contraction depending upon the shape of the fillers, whether they are fibrous or spherical, the amount added in weight percent, and their coefficients of contraction. Fibrous fillers (glass fibers) produce the greatest reduction in contraction in the direction perpendicular to the molding pressure; spherical fillers (bronze powder) tend to

equalize contraction in both directions[8]. The laminated-glass-cloth-with-Teflon-binder material has the greatest contraction in the direction of the glass cloth layers. Filled Teflon cages are usually made from molded tube stock with the molding pressure direction parallel to the axis of the tube (cage). The laminated-Teflon cages are also made from tube stock. The tube is made by continuously wrapping glass cloth on a mandrel and binding the successive layers together with Teflon resin.

The total contraction from 70 to -423° F of several filled Teflon and the laminated-glass-cloth Teflon materials are given in table II. The materials shown are similar to cage materials discussed previously. It can be noted that the glass-fiber-filled and the laminated-glass-cloth Teflon materials exhibit anisotropic contraction with respect to the cage radial and width directions. The radial direction is perpendicular to direction of molding pressure for the glass-fiber-filled material and through the glass cloth layers for the laminated-glass-cloth material, as noted in table II. The bronze-filled Teflon has the same contraction in both directions. Contraction in the radial direction will decrease the cage locating clearances, for inner-race located cages, whereas contraction in the width direction will decrease ball-pocket clearances. With filler materials or glass cloth layers, the total contraction of the Teflon cage materials shown in table II, range from two to eight times greater than that for the 440C ball and race material.

Typical locating-race and ball-pocket clearances for seven cage materials are shown in table III. The first four cages listed are fabricated from Teflon based materials and have contraction properties similar to those described above. Their inner-race locating clearances range from 0.017 to 0.038 inch and the ball-pocket clearances from 0.014 to 0.026 inch.

The relative contraction of cage reinforcing materials such as aluminum and stainless steel can also affect cage clearances at cryogenic temperatures. The total contraction for 2024 aluminum is about 0.0042 inch per inch, whereas that for AISI 410 or 416 stainless steel is approximately 0.0019 inch per inch, from 70 to -423° F[10], which is the same as the 440C ball and race material.

In table III bearing cages using materials 1, 3, and 7 were reinforced with aluminum. Cages fabricated from materials 5 and 6 used stainless steel shrouds. A glass-fiber-filled Teflon cage (material 3, table III) supported with a riveted aluminum shroud experienced ball pocket cracking after running about 8 hours in -400° F hydrogen gas. The cracking was partly caused by the different contraction rates between the glass-fiber-filled Teflon body (0.0164 in./in.) and the aluminum shroud (0.0042 in./in.). A more complete analysis of the cage failure is discussed in the following section.

Cage failure mechanisms. - An example of a brittle failure of a cage material (material 7, table III) is illustrated in figure 13(a). The molybdenum disulphide-(MoS_2) filled-polyimide cage experienced complete failure after running only 22 minutes in hydrogen gas at -400° F.

The cage design was similar to that shown in figure 11(c).

When an inner-race located cage is cooled to cryogenic temperature, insufficient clearance at the locating race will cause binding and the excessive heat generated at the surface may lead to cage failure. Bearing 23-S in table III is an example of a cage with insufficient clearance. In the initial run the bearing was brought to a complete stop from 5000 rpm with a 200 pound applied load.

The combination of a larger-than-required inner-race clearance and poor wear resistance can also result in cage failure. A silver composite cage (bearing 11-S, table III) fitted with a stainless steel shroud experienced this type of failure. The bearing had run in -400° F hydrogen gas at 20 000 rpm and 200 pounds for 138 minutes. The bearing after test is shown in figure 13(b). During bearing operation the shroud had rubbed on the outer-race land and moved relative to the cage body. The balls wore into the shroud and jammed the bearing.

Although a cage material may have good wear resistance, failure can result from structural deficiency in the material even when reinforcing is used. The delamination of the laminated-glass-cloth-with-Teflon-binder cage material is shown in figure 13(c). After running 349 minutes, this cage delaminated between two ball pockets. Delamination probably resulted from insufficient Teflon binder between the cloth layers, or from an improper curing technique during lamination.

The glass-fiber-filled Teflon cage material reinforced with a riveted aluminum shroud showed structural deficiency of another type. After running 464 minutes, cracks appeared in a ball pocket

of the cage (fig. 13(d)). It is speculated that the cracks resulted from the difference in contraction rates between the Teflon cage body and the aluminum shroud. The body shrank away from the shroud when the bearing was cooled to -400° F. When the bearing was subsequently run at high speed, centrifugal growth of the body between rivets caused fracture of the cage at the thin web section. Subsequent runs were made in -400° F hydrogen gas with other bearings using glass-fiber-filled Teflon cages without aluminum shrouds. No pocket cracking occurred in these cages when the bearings were run at speeds to 40 000 rpm.

SUMMARY

Rolling element bearings used in rocket engine turbopumps have specific lubrication, material, and design requirements. The bearing load-carrying surfaces are lubricated by thin transfer films provided by a self-lubricating cage material, which is usually a Teflon compound. The following factors are emphasized for a good bearing design:

1. Minimum heat generation at high operating speed can be obtained by designing the bearing with open race curvatures and small ball diameters.

2. The formation and life histories of transfer films on the bearing load-carrying surfaces must be sufficient to provide bearing lubrication for several hours without any race wear. The lubricating capability and wear of several filled-Teflon cage materials have been evaluated for bearing running times up to 10 hours.

3. In addition to the self-lubricating feature, the cage must have adequate strength and be designed with proper clearances to prevent seizure or excessive wear that can result in premature bearing failure.

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TABLE I. - COMPARISON OF BALL SPIN TORQUES OF TWO 40 mm BALL BEARINGS. TORQUE ABOUT BEARING AXIS - LB-IN.

$T = 100$ lb, $N = 40\,000$ rpm, $\beta' = 10^{\circ}$, $f = 0.56$

Race Curvatures	$\rho_1 = 0.51$, $\rho_o = 0.58$	$\rho_1 = 0.52$, $\rho_o = 0.54$
108 Series $d = 0.375$ in.	2.095	1.653
1908 Series $d = 0.250$ in.	1.831	1.412

TABLE II - TOTAL CONTRACTION OF TFE TEFLON MATERIALS FROM 70 to -423° F (from Ref. 10)

Material (weight percent)	Total Contraction in cage radial direction (1) (in./in.)	Total Contraction in cage width direction (2) (in./in.)
100% Teflon	0.0215	Same as radial direction
15% Glass fibers 85% Teflon	0.0084 (3)	0.0164 (3)
25% Glass fibers 75% Teflon	0.0090	0.0165
65% Bronze 35% Teflon	0.0140	Same as radial direction
38% Glass cloth 62% Teflon	0.0135	0.004

(1) Radial direction is perpendicular to molding pressure and through glass cloth layers.

(2) Width direction is parallel to molding pressure and glass cloth layers.

(3) From manufacturer's data

TABLE III - TEST-BEARING CAGES (ref. 2)

(Deep-groove ball bearings, 40-mm bore, separable at outer race; races and balls, AISI 440C stainless steel; number of balls, 10; ball diameter, 0.375 in. (0.953 cm); inner- and outer-race curvature, 0.54; radial clearance, 0.0025 in. (0.0064 cm).)

Cage	Approximate weight percent of materials	Cage construction	Bearing	Inner land clearance		Ball pocket clearance	
				in.	cm	in.	cm
1	38 Percent glass cloth laminates with 62 percent Teflon binder[a]	One-piece body with riveted aluminum side plates	13-S	0.018	0.046	0.019	0.048
			16-S	0.017	0.043	0.015	0.038
2	15 Percent glass fibers, 5 percent molybdenum disulfide, 80 percent Teflon[a]	One-piece body with no external support	14-S	0.031	0.079	0.014	0.036
			23-S	[b]0.048	[b]0.122	0.018	0.046
3	15 to 20 Percent glass fibers. balance Teflon[a,c]	One-piece body with one-piece riveted aluminum shroud	15-S	0.035	0.089	0.016	0.041
			22-S	0.035	0.089	0.026	0.066
4	30 Percent bronze powder, 70 percent Teflon[a]	One-piece body with no external support	20-S	0.038	0.097	0.018	0.046
			21-S	[d]0.019 to 0.066	[d]0.048 to 0.168	0.016	0.041
5	78 Percent copper, 9 percent Teflon, 13 percent tungsten diselenide[e]	Shrink-fit one-piece stainless-steel shroud over one-piece body pinned in two places 180° apart	17-S	0.027	0.069	0.015	0.038
6	85 Percent silver, 5 percent Teflon, 10 percent tungsten diselenide[e]	Same as cage 5 material without 2 pins	11-S	0.024	0.061	0.015	0.038
		Same as cage 5 material	18-S	0.025	0.064	0.015	0.038
7	85 Percent polyimide, 15 percent molybdenum disulfide[a]	Shrink-fit one-piece aluminum shroud over one-piece body	12-S	0.022	0.056	0.016	0.041
		Shrink-fit one-piece aluminum shroud over one-piece body with 2 pins 180° apart	19-S	0.019	0.048	0.016	0.041

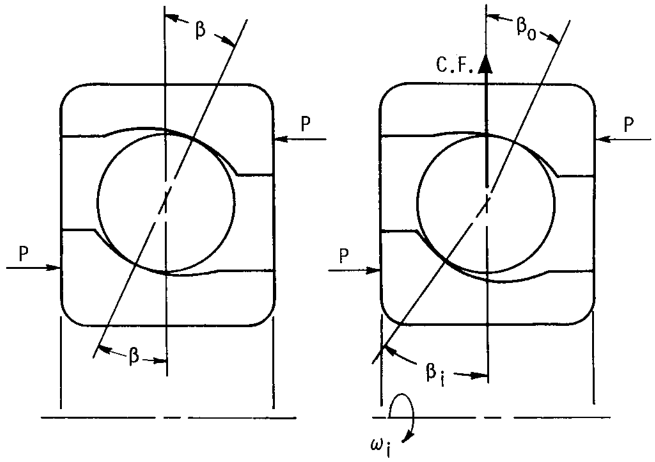
[a]Manufacturer's data.

[b]Machined to this clearance after running one test with 0.023-in. (0.058 cm) clearance .

[c]Less than 1 percent ferric oxide added as coloring agent.

[d]Inner diameter of cage machined eccentric.

[e]Metal composites weight percent calculated from measured specific gravity values.



(a) Contact angle when under load. (b) Contact angles when under load and at high speed.

Figure 1. - Effect of high shaft speed on bearing contact angle.

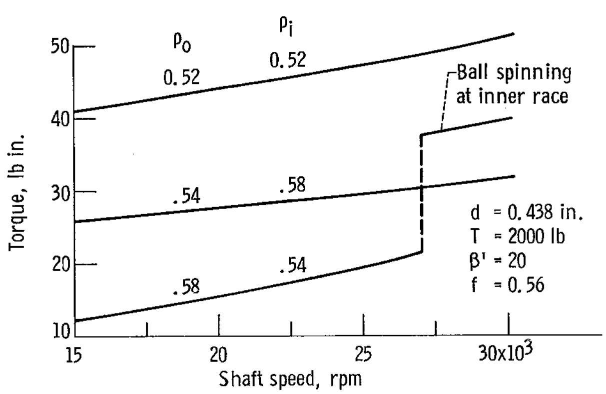


Figure 2. - Torque due to ball spinning for 50 millimeter (210 series) bearings.

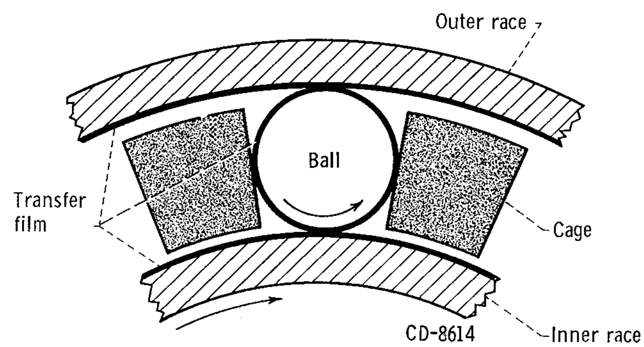
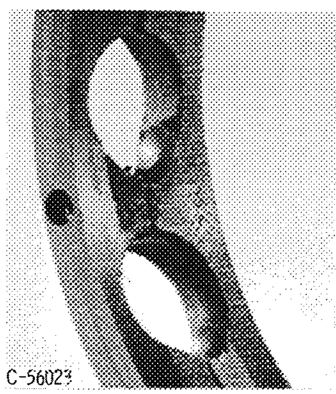
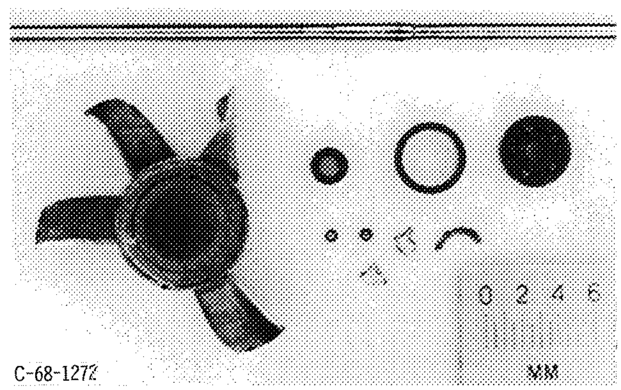


Figure 3. - Film transfer mechanism.

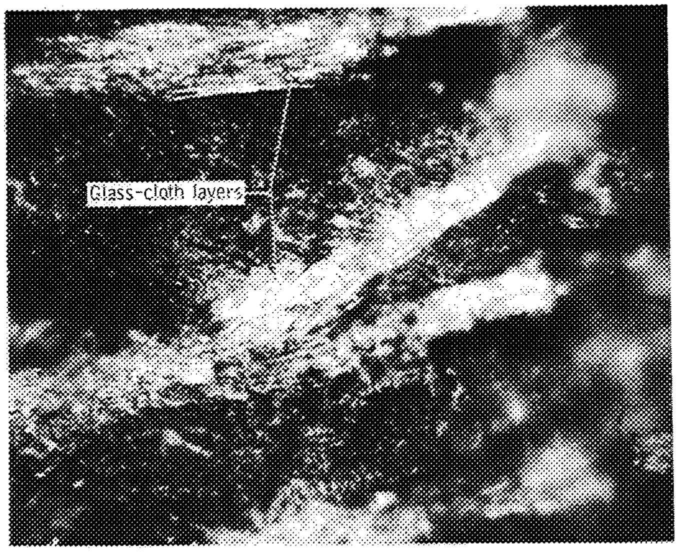




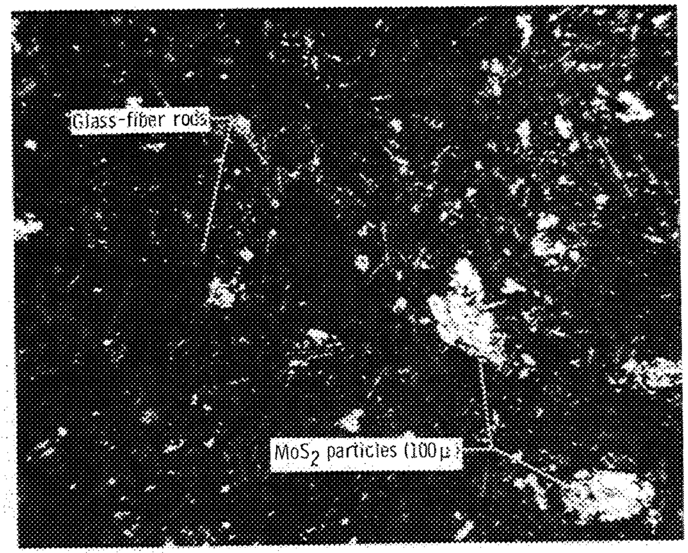
(a) Extreme ball-pocket wear after 80 minutes. 40 mm ball bearing with cotton-cloth phenolic cage. Speed, 10 000-40 000 rpm.

(b) Complete cage failure after 90 minutes. Turbine flow meter bearing (0.047 inch bore X 0.156 inch O.D.) with steel cage of "L type" design. Speed, 1000-9000 rpm.

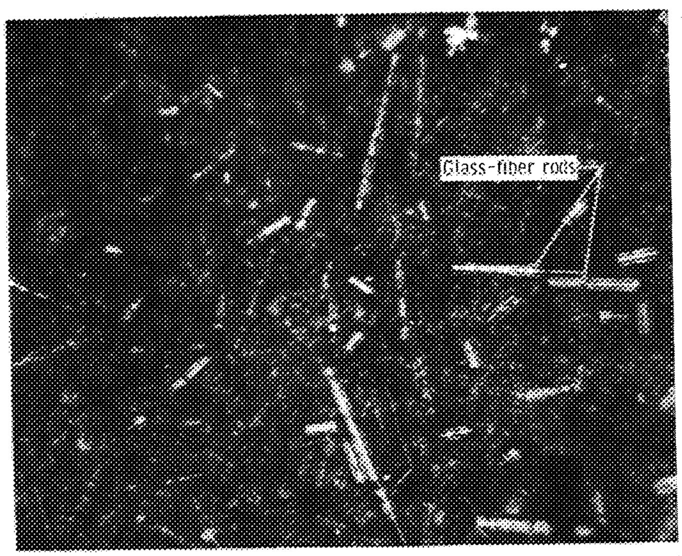
Figure 4. - Bearing cage failures in liquid hydrogen.



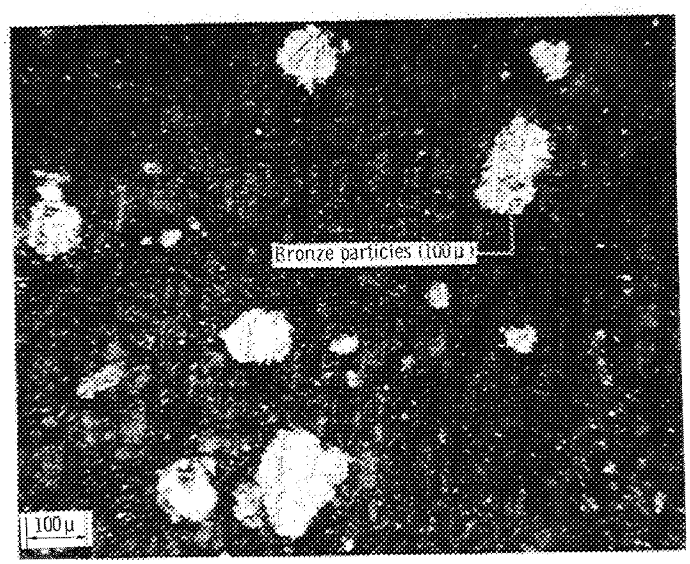
(a) Cage material 1, 62 percent Teflon, 38 percent glass cloth.



(b) Cage material 2, 80 percent Teflon, 15 percent glass fibers, 5 percent MoS_2.



(c) Cage material 3, 85 percent Teflon, 15 percent glass fibers.



(d) Cage material 4, 70 percent Teflon, 30 percent bronze powder.

Figure 5. - Structure of four self-lubricating cage materials.

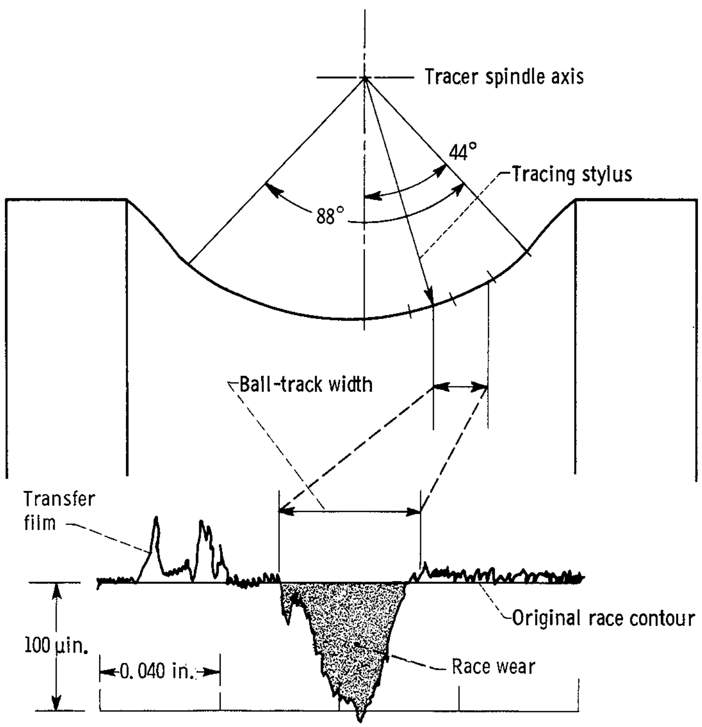


Figure 6. - Profile trace of bearing inner race normal to ball-rolling direction.

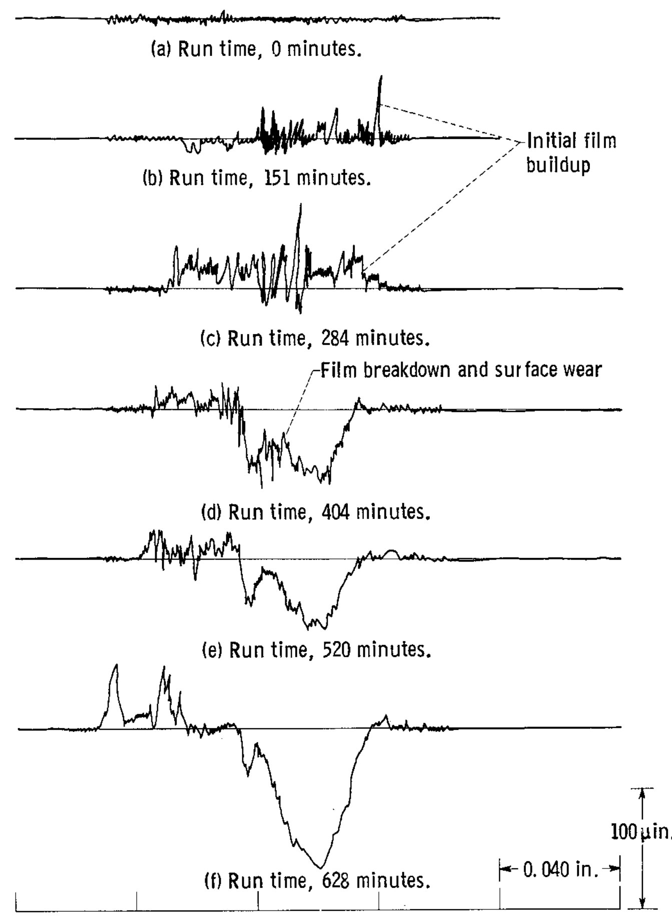


Figure 7. - Progressive profile traces of inner-race groove (normal to ball-rolling direction). Cage material, 38 percent glass-cloth with 62 percent Teflon binder; shaft speed, 20 000 rpm; thrust load, 200 pounds; coolant, hydrogen gas at 60° R.

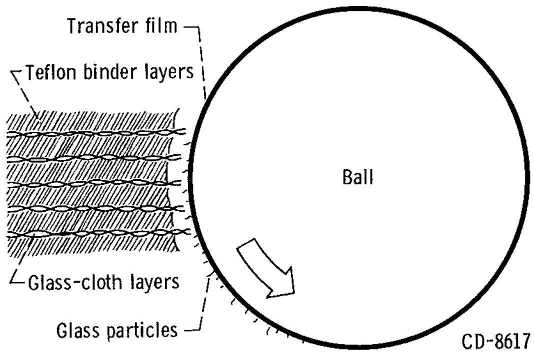
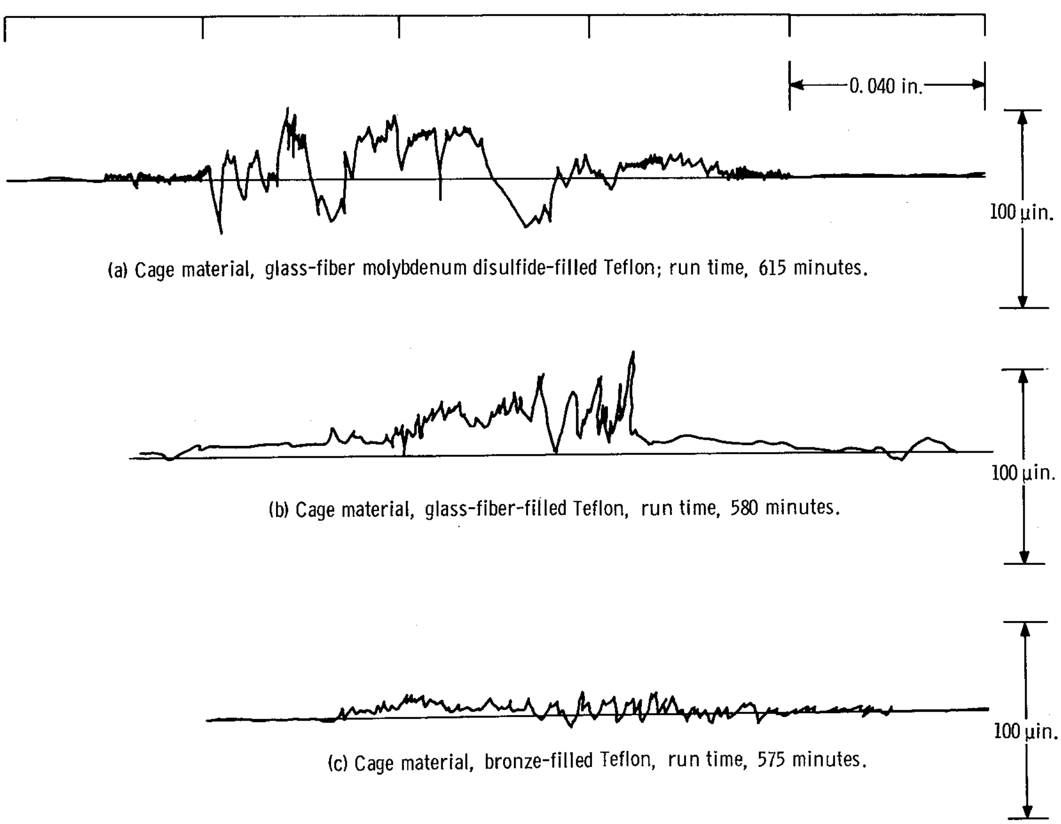


Figure 8. - Postulated wear process in ball pocket of glass-cloth-with-Teflon-binder cage.

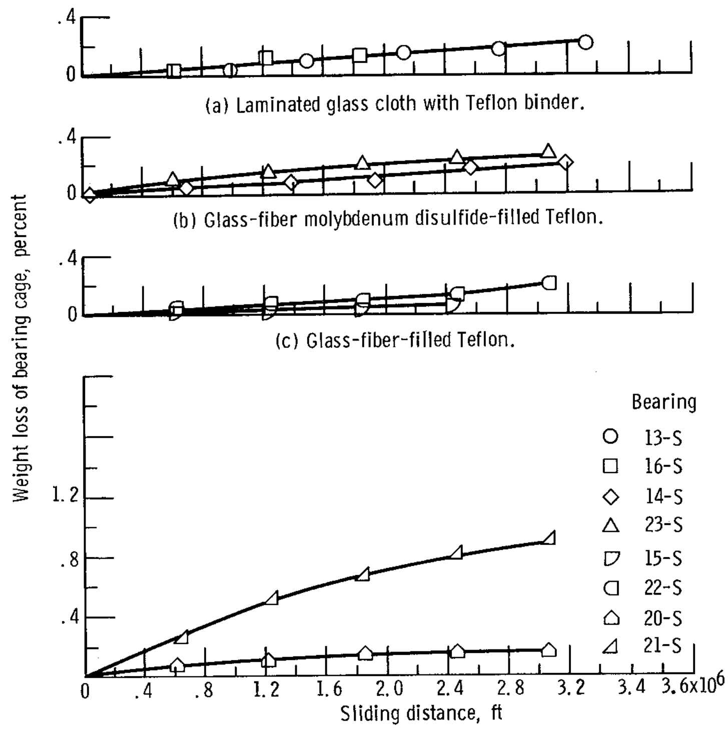


(a) Cage material, glass-fiber molybdenum disulfide-filled Teflon; run time, 615 minutes.

(b) Cage material, glass-fiber-filled Teflon, run time, 580 minutes.

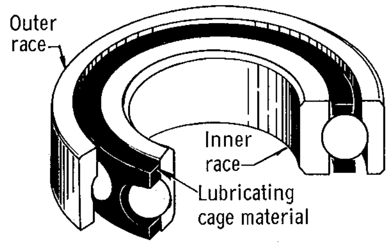
(c) Cage material, bronze-filled Teflon, run time, 575 minutes.

Figure 9. - Profile traces of inner-race grooves for bearings run with filled-Teflon cage materials. Shaft speed, 20 000 rpm; thrust load, 200 pounds; coolant, hydrogen gas at 60° R.

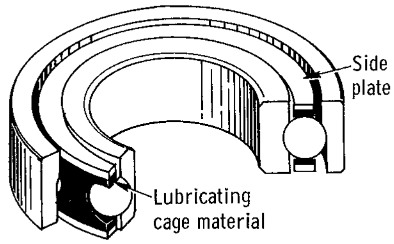


(d) Bronze-filled Teflon.

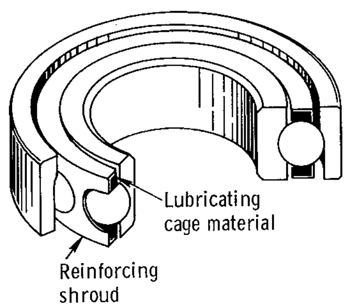
Figure 10. - Bearing cage weight loss as a function of sliding distance relative to inner race. Shaft speed, 20 000 rpm; thrust load, 200 pounds; coolant, hydrogen gas at 60° R.



(a) One-piece body with no external support.

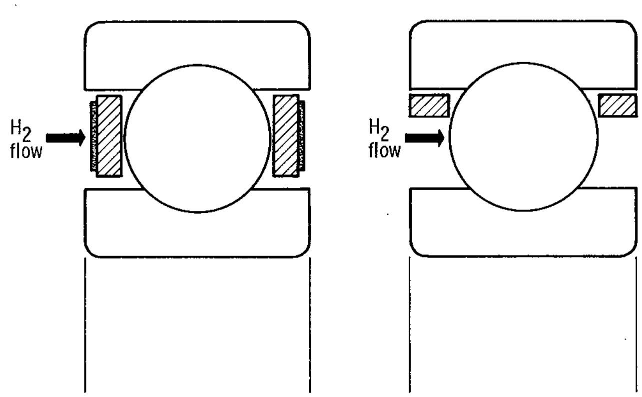


(b) One-piece body with side plates.



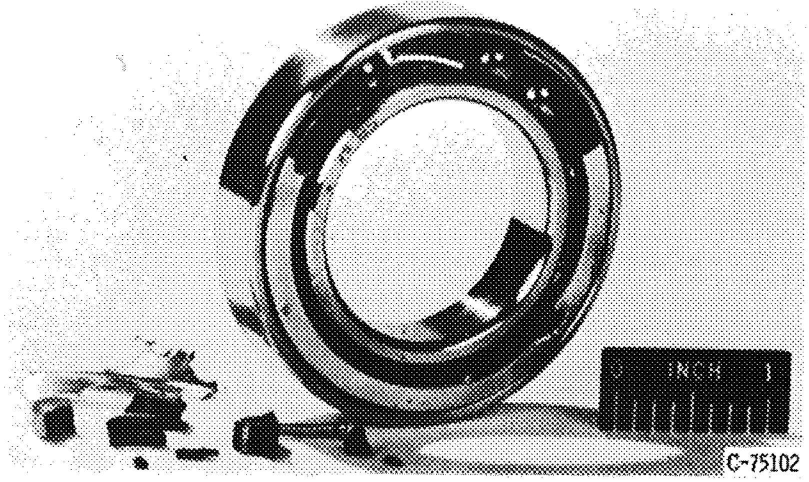
(c) One-piece body with one-piece shroud.

Figure 11. - Inner-race located cage designs.

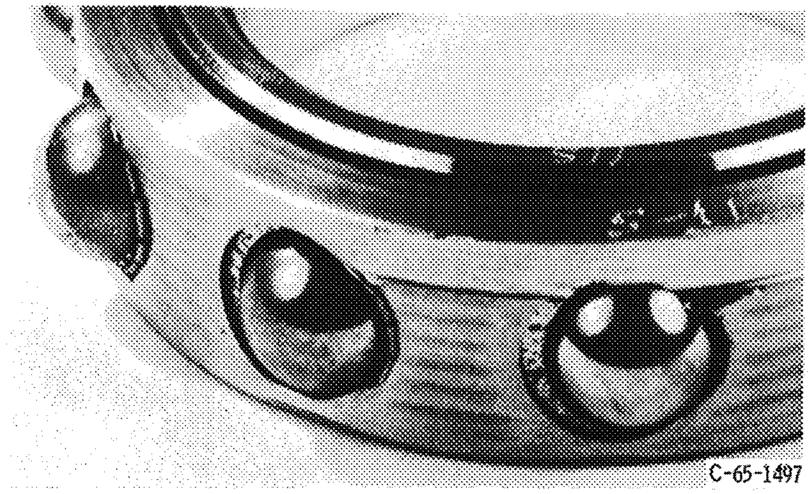


(a) Conventional design. (b) Thin-line design.

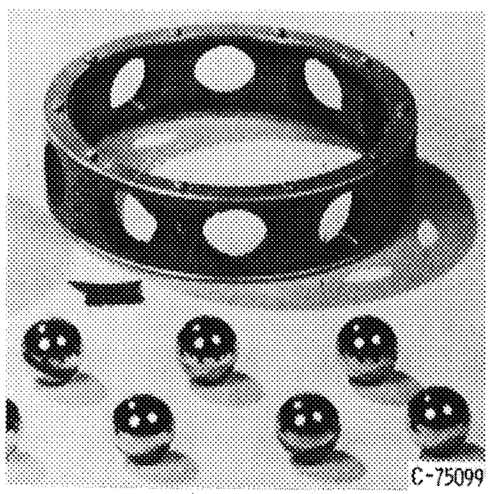
Figure 12. - Outer-race located cage designs.



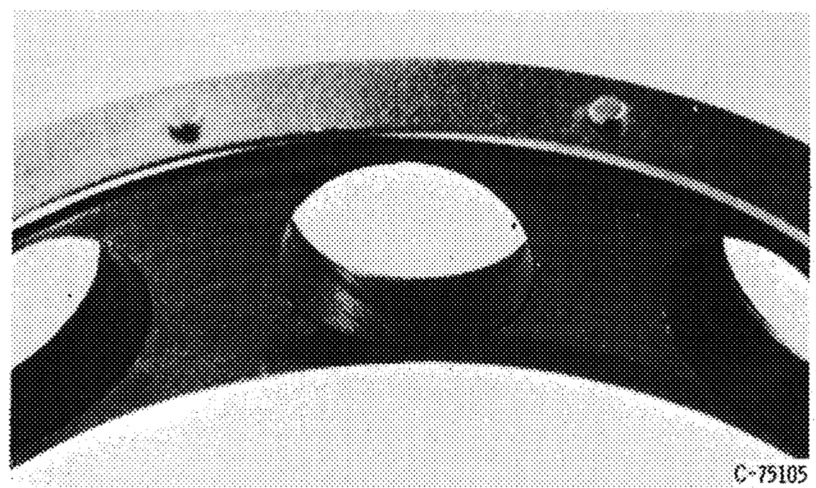
(a) Molybdenum disulfide-filled polymide. Complete failure after 22 minutes.



(b) Silver composite. Shroud rubbed on outer-race land and moved relative to body; ball wore into shroud; bearing jammed after 138 minutes.



(c) Laminated glass cloth with Teflon binder. Delamination after 349 minutes.



(d) Glass-fiber-filled Teflon. Cracks in ball pocket after 464 minutes.

Figure 13. - Mechanical damage of bearing cages. Shaft speed, 20 000 rpm; thrust load, 200 pounds; coolant, hydrogen gas at 60° R (-400° F).

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